PROGRESS REPORT

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Vibrations and Structureborne Noise in Space Station

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1. Summary of Major Activities

During the reporting period, the following major activities relating to the proposed work have been accomplished.

1.1 Conference Papers (Presented)

1. Vaicaitis R. and Mixson J.S., "Review of Research on Structureborne Noise," 26th AIAA/ASME/ASCE/AHS SDM Conference, Paper No. 85-0786-CP, Orlando, Fl., April, 1985.
2. Vaicaitis, R. and Bofilios, D.A., "Response of Double Wall Composite Shells," 26th AIAA/ASME/ASCE/AHS SDM Conference Paper No. 85-0604-CP, Orlando, Fl., April, 1985.

1.2 Conference Papers (Prepared)

1. Vaicaitis, R. and Bofilios, D.A., "Noise Transmission of Double Wall Composite Shells," ASME 10th Biennial Conference on Mechanical Vibration and Noise, Paper No. H-334, Cincinnati, Ohio, September 1985.
(A copy of this paper is included with the present progress report)

1.3 Oral Presentations

1. Vaicaitis, R. and Bofilios, D.A. "First Progress Report on Structureborne Noise for NASA NAG-1-541 Grant," Langley Research Center, NASA, ANRD, March 25-27, 1985.
2. Vaicaitis, R., "Second Progress Report for NASA NAG-1-541 Grant," Langley Research Center, NASA, ANRD, Aug. 5-7, 1985.

2.0 Technical Highlights

The technical background on structureborne noise generation and transmission for aircraft, rotorcraft, automobile, spacecraft, ship and building technology has been described in Ref Review of technical literature related to this subject suggests that theories on structureborne noise generation, propagation and transmission are incomplete and significant amount of theoretical and experimental work is needed before noise control measures can be implemented. The analytical techniques which are emerging as potential candidates for analysis of structureborne noise are modal methods, finite element procedures and wave propagation techniques. However, the success and validity of using those methods is strongly influenced by the ability to describe in detail input loads, vibrational energy propagation through complex structural systems and coupling of acoustic field to structural vibrations. The first phase of the proposed work during the 1984-85 period was mainly devoted in reviewing the related literature and then developing preliminary analytical model for simplified acoustic and structural geometries for pressurized and unpressurized Space Station modules. In addition to the analytical work, an experimental program on structureborne noise generation and transmission was started. In what follows, a brief review of those accomplishments is given.

2.1 Theoretical Study of Structureborne Noise Transmission

Analytical models based on modal analysis have been developed for a cylindrical acoustic enclosure shown in Figs. 1 and 2. Single wall (Fig. 1) and double wall (Fig. 2) constructions have been considered. The shells are

closed at both ends by circular plates of either single or double wall constructions. The forces acting on the structure are normal point loads which can be located either on the shell or on the end plates (Fig. 3). The point loads are random and can be acting at any arbitrary location on the structure (inside and/or outside). Numerical procedures have been developed to calculate modal frequencies, deflection response and noise transmission. Pressurized and unpressurized modules were considered. The geometric parameters selected in the first phase of this study are similar to those of the proposed Space Station modules. Structural models for aluminum and fiber reinforced composite materials were developed for both single and double wall shells. The details of the theoretical development are given in Refs. - . The highlights of the numerical results are presented in this proposal.

2.1.1 Single Wall Aluminum Shell

The aluminum shell shown in Fig. 1 has the following dimensions: $L = 300$ in, $R = 58$ in and $h_s = 0.1$ in. Both ends are closed and the interior walls are lined with a layer of porous acoustic material [3,4]. The inputs to the shell located at $x_1=x_2 = 150$ in., $\theta_1 = -90^0$, $\theta_2 = 90^0$ are random point loads which are characterized by a truncated Gaussian white noise spectral density [3,4]. The shell response calculated at $x = L/2$, $\theta = 45^0$ is shown in Fig. 4 for pressurized and unpressurized conditions. The deflection response levels RL are calculated from

$$RL\ (x,\theta,\omega) = 10 \log [S_w(x,\theta,\omega)\Delta\omega/w_{ref}^2] \qquad (1)$$

where S_w is deflection response spectral density, $\Delta\omega$ is frequency bandwidth and $w_{ref} = h_s$. As can be observed from these results, pressurization has a

marked effect on shell response at low frequencies for the first few shell modes. Similar results are presented in Fig. 5 where sound pressure levels inside the shell generated by two point loads are given. These results suggest that pressurization plays only a minor role on noise transmission. It should be noted that under orbital conditions noise can be generated only for the pressurized shell while vibrations are important for both cases. The results shown in Figs. 4 and 5 indicate that shell response is dominated by low frequency structural resonances while transmitted noise is governed by structural and acoustic resonances in the mid-frequency range (300 Hz-700 Hz). This can be attributed to the fact that low frequency shell modes do not couple well to the low frequency acoustic modes.

2.1.2 Single Wall Composite Shell

The composite shell is constructed from 10 laminae layers each reinforced by fiberglass and/or graphite fibers. Each layer can be oriented in any arbitrary direction. The details of the theoretical development and parameters chosen to characterize composite materials are given in Refs. 3-6. Fiberglass and graphite fibers are used to reinforce the plexiglass material. The ratio of fibers volume to the plexiglass volume is 0.2. The weight of the composite shell is about half of the weight of the aluminum shell. The transmitted noise for an aluminum and composite shells is given in Fig. 6. These results tends to indicate that more noise is transmitted at most frequencies by a shell made from composite materials. It should be noted that structural damping mechanisms used to characterize vibration attenuation are different for these two cases.

2.1.3 Double Wall Aluminum Shell

Deflection response and noise transmission of a double wall shell shown in Fig. 2 was calculated for point load inputs. For Space Station applications, the outer shell could serve as a radiation and thermal protection shield. The space between the two shells is filled with soft thermal insulating material. The vibration coupling between the two shells is provided by the soft thermal material. Numerical results are presented for a core with h_s^c = 2 in (thickness), k_s = 4.16 lb/in^3 (uniaxial stiffness), and ρ_s = 3.4×10^{-6} lb - sec^2/in^4 (density).

The natural frequencies of a double wall aluminum shell are plotted in Fig. 7 for 10 longitudinal and 20 circumferential modes. The thicknesses of the outer and the inner shells are h_E = 0.032 in and h_I = 0.1 in. For the present double wall construction, the flexural (in phase) and the dilatational (out of phase) modes are included. The highest modal frequency is that of "breathing" mode for which n = 0 and m = 1 (simply supported shell at both ends). Results plotted in Fig. 7 indicate that for the large shell dimensions and the ratio radius/length = 0.1933 chosen in this study to characterize the dimensions of a space station module the modal frequencies at n = 0 seem to converge to a single point for all values of m = 1,2,...,10. This suggests that in the vicinity of the "breathing" mode frequency large number of structural modes could couple tc acoustic modes resulting in high levels of noise transmission. In Fig. 8, sound pressure levels in the shell generated by two mechanically induced point loads are given for reverberant (hard walls) and absorbent (interior walls lined with porous acoustic materials) conditions. For reverberant conditions, the noise levels inside the cylinder become relatively large and are dominated by peaks at acoustic resonant frequencies.

2.1.4 Double Wall Composite Shell

The modal frequencies of a double wall composite shell constructed from ten laminae layers for inner shell and three laminae layers for the outer shell are shown in Fig. 9. From the results given in Figs. 7 and 9, it can be seen that modal frequencies of a composite shell are significantly higher than those of an equivalent aluminum shell. However, the mass of the composite shell is about 50% less than that of the aluminum shell. Figure 10 depicts sound pressure levels for an aluminum and fiber reinforced composite shell for identical loading and damping conditions. As can be observed from these results, the noise levels generated by a composite shell are higher than the noise levels for an aluminum shell at most frequencies. However, the composite shell is much stiffer than the aluminum shell. A shift in modal frequency could induce different coupling between structural and acoustic modes. Since damping mechanisms of composite materials are significantly different from those of metals, it is difficult to assess the equivalence between composites and metals. The response levels of the inner composite shell are shown in Fig. 11 for several values of the loss factor g_{ij}. In this case, all the stiffness moduli are complex with $E_{ij} = \overline{E}_{ij} (1 + \underset{\sim}{i}\, g_{ij})$, $G_{ij} = \overline{G}_{ij} (1 + \underset{\sim}{i}\, g_{ij})$, etc., where $\overline{E}_{ij}$ and $\overline{G}_{ij}$ are real quantities. These results indicate that large amount of response reduction can be achieved in a composite shell for large values of loss factor g_{ij}. The loss factor g_{ij} is function of matrix material, ratio of fibers to material, fiber orientation, number of laminae layers, etc. The results presented in Fig. 12 correspond to point loads acting on the interior shell at $x_1^i = x_2^i = L/2$, $\theta_1^i = -90^0$ and $\theta_2^i = 90^0$. The fiber orientation of the three layers (Fig. 2) at the exterior shell is described in Fig. 12. The fiber orientation for the ten layers of the interior shell are (A) 0°,22.5°,45°,45°.22.5°,0°,90°,90°,90°,90° (B) 90°,

0°,90°,0°,90°,0°,90°,0°,90°,0° (C) -45°,45°,-45°,45°,-45°,45°,-45°,45°,-45°, 45°. These results show that interior noise is a function of fiber orientation in a composite shell. The interior noise levels might be tailored to meet specific needs by selecting a suitable fiber orientation. However, interior noise is a function of frequency and only specific frequency bands might be affected by this procedure.

2.1.5 Double Wall Circular End Plates

Analytical models for deflection response and noise transmission of double wall circular aluminum plates (Figs. 2,3) to point loads were developed. The details of the theoretical analysis are given in Ref. 3. The response levels of the outer and inner plates for point loads acting on the outer plate are shown in Figs. 13 and 14. The thicknesses of the outer and the inner plates are identical and equal to 0.25 in. The response levels are calculated at $r = 0$ and $\theta = 45^0$. As can seen from these results, response levels are significantly higher at most modal frequencies when the point loads are located at $r_1^T = r_2^T = 10$ in. and $\theta_1^T = 0^0$, $\theta_2^T = 180^0$. The sound pressure levels at $x = L/2$, $r = 23$ in. and $\theta = 45^0$ due to noise transmitted through double wall circular end plates located at $x = L$, are shown in Fig. 15. The inputs are two point loads located at $r_1^T = r_2^T = 28$ in and $\theta_1^T = -90^0, \theta_2^T = 90^0$. For comparison, noise transmitted through a double wall aluminum shell is included in this figure. It can be seen that interior noise is dominated by end plate vibrations for frequencies up to 200 Hz and by shell vibrations for frequencies above 200 Hz. These results indicate that reglecting noise transmitted by the end caps would underestimate interior sound pressure levels for the low frequency region.

2.2 Experimental Study of Structureborne Noise

The basic objective of the experimental study is to assist in the fundamental understanding of generation and transmission of structureborne noise. To achieve these goals funds from the Department of Civil Engineering and Engineering Mechanics were allocated to construct laboratory facilities and to purchase additional vibration and noise measurement equipment. The following equipment and computer programs have been acquired: ZONIC four channel real-time spectrum analyzer, monochrome and color monitors, dot matrix plotter, 14 channel FM tape recorder, modaly tunned hammer, miniature accelerometers and signal conditioners, acoustic emission equipment, electromagnetic shaker and power amplifier, MODAL and MODAL MODIFICATION software. Series of preliminary tests have been conducted utilizing the AeroCommander aircraft fuselage. The results shown in Figs. 16 and 17 indicate typical interior noise levels generated by a shaker excitation to the sidewall (structureborne) and by exterior noise from two speakers (airborne) located at about 3 ft. from the fuselage. The results presented in Fig. 16 correspond to interior point located very close to the vicinity where the shaker excitation is applied. The results shown in Fig. 17 are for an interior point located at about 8 ft. from the shaker. In the vicinity of point load excitation, interior noise is dominated by the structureborne contribution. Except for the low frequency region of 40Hz-80Hz, the magnitude of structureborne noise decreases rapidly with increasing distance from the point of mechanical excitation. For this aircraft, the natural frequencies of the main structural frame vibrations accur in the frequency range of 40 Hz-80Hz. These preliminary results tend to suggest that at some distance away from source location structureborne noise is mainly transmitted by low frequency vibrations of the main frame structure.

3.0 Future Work

We expect to continue the development and improvements of the analytical models for application to response and noise transmission estimation for space station applications. In addition, experiments will be conducted on structure-borne noise generation, propagation and transmission.

4.0 References

1. Vaicaitis, R., and Mixson, J.S., "Review of Research on Structureborne Noise" 26th AIAA/ASME/ASCE/AHS SDM Conference, Paper No. 85-0786-CP, Orlando, Fl., April, 1985.

2. Vaicaitis, R., and Bofilios, D.A., "Response of Double Wall Composite Shells," 26th AIAA/ASME/ASCE/AHS SDM Conference, Paper No. 85-0604-CP, Orlando, Fl., April, 1985.

3. Bofilios, D.A., "Response and Noise Transmission of Double Wall Circular Plates and Laminated Composite Cylindrical Shells," Doctoral Thesis, Columbia University, New York, N.Y., July, 1985.

4. Vaicaitis, R. and Bofilios, D.A., "Noise Transmission Through Double Wall Composite Shells," 10th Biennial Conference on Mechanical vibration and Noise, ASME, Volume H-334, Cincinnati, Ohio, Sept., 1985

5. Bert, C.W. and Eagle, M.D., "Dynamics of Composite, Sandwich, and Stiffened Shell-Type Structures," Journal of Spacecraft and Rockets, Vol. 6, No. 12, Dec. 1969, pp. 1345-1361.

6. Soedel, W., Vibrations of Shells and Plates, Marcell Dekker Inc., 1981.

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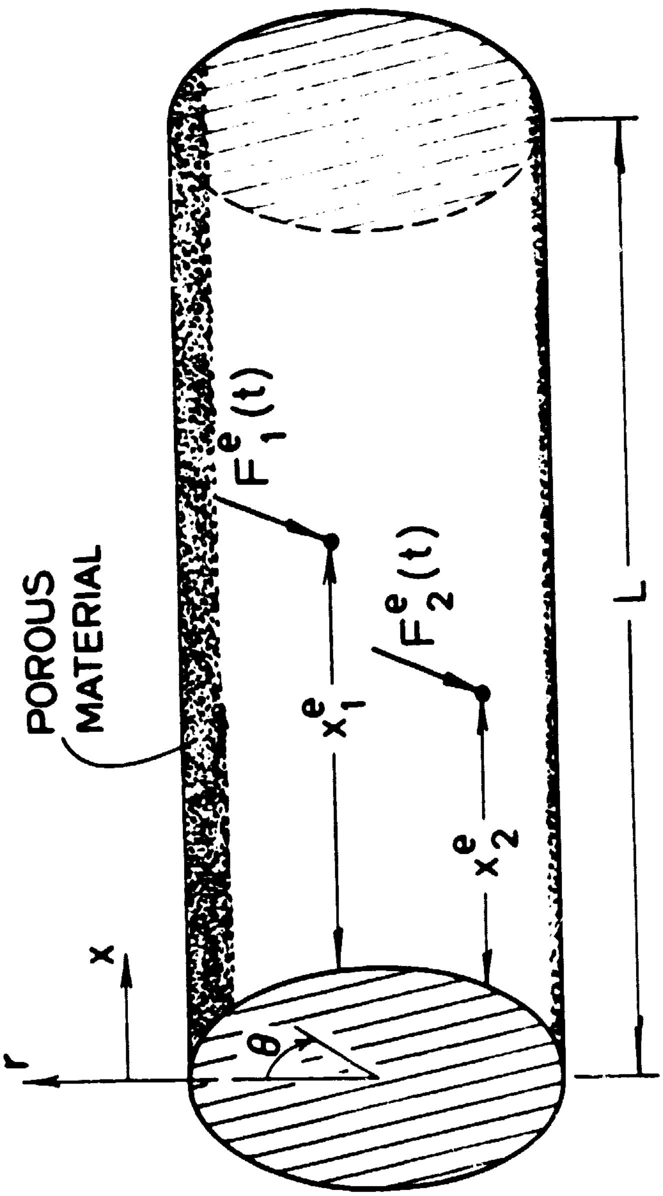
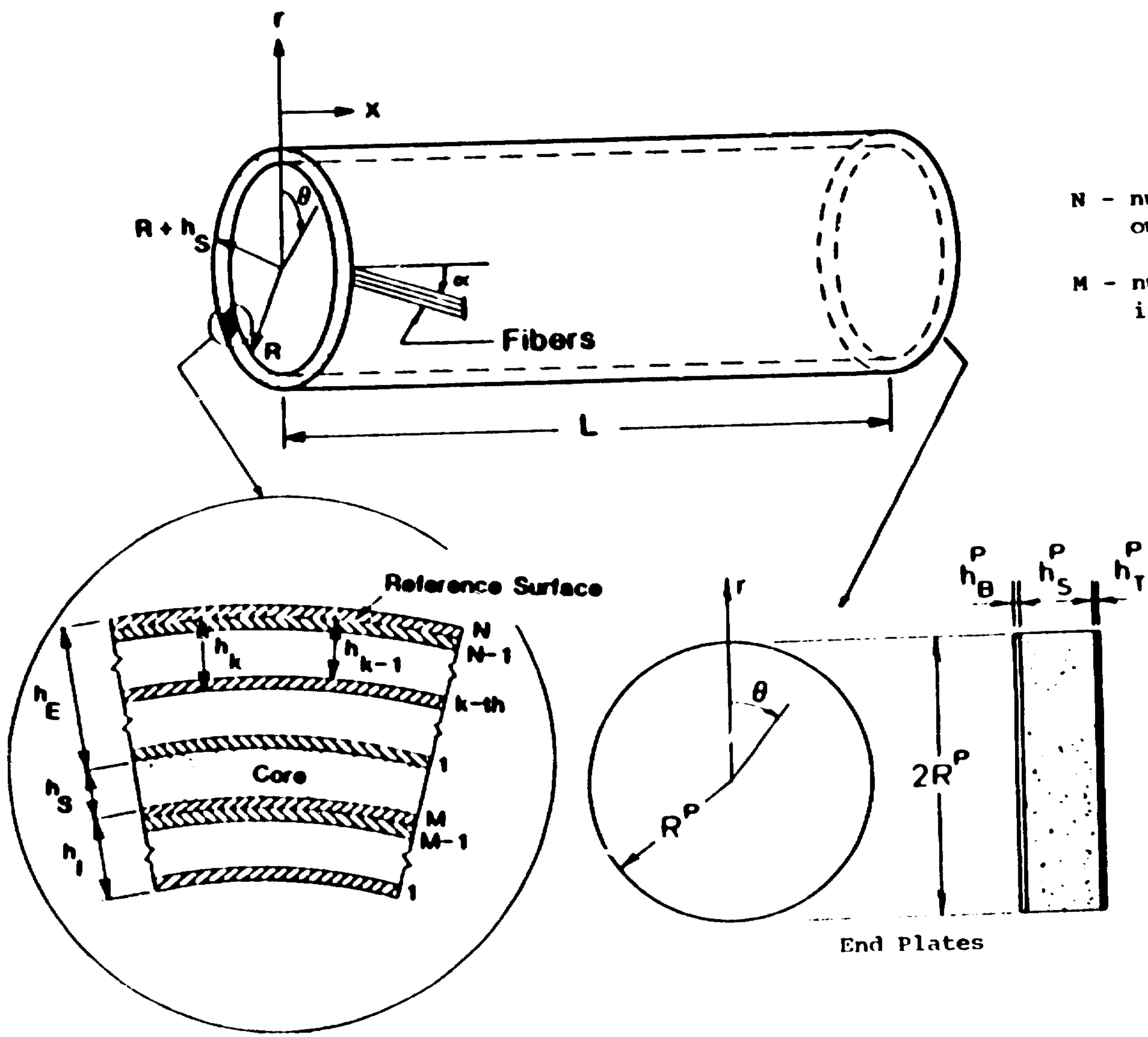


Fig. 1 Problem Geometry of a Cylindrical Shell



N - number of laminae layers of outer shell

M - number of laminae layers of inner shell

Figure 2. System's Geometry

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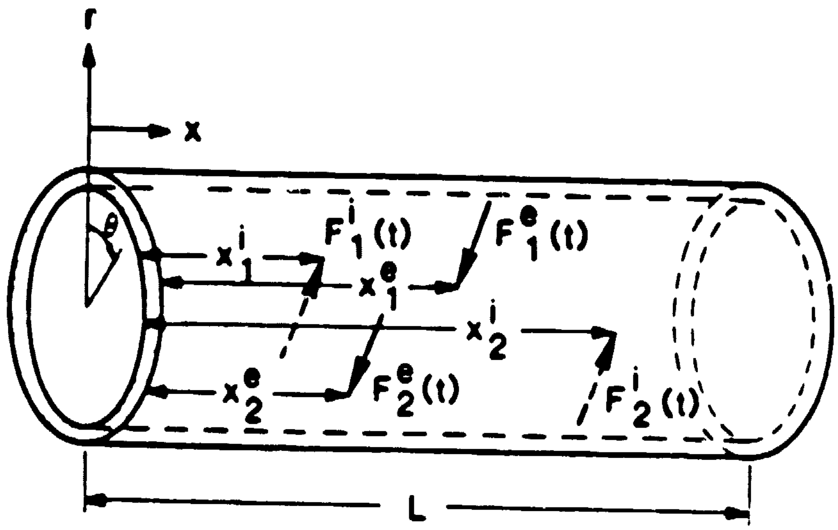

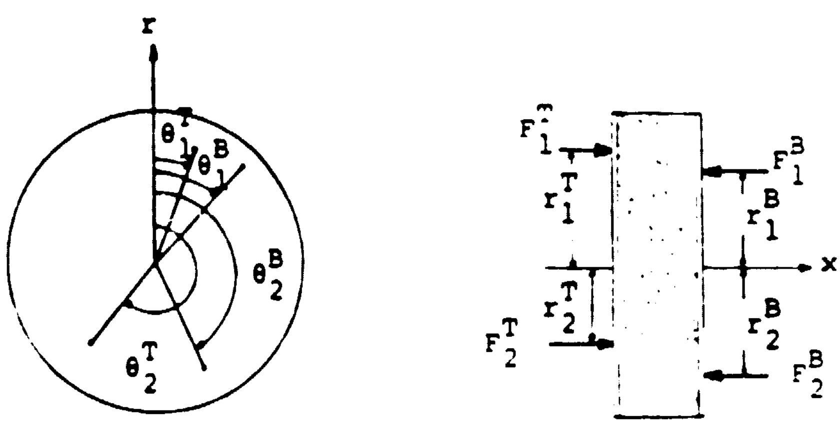

Figure 3. Location of Random Point Loads.

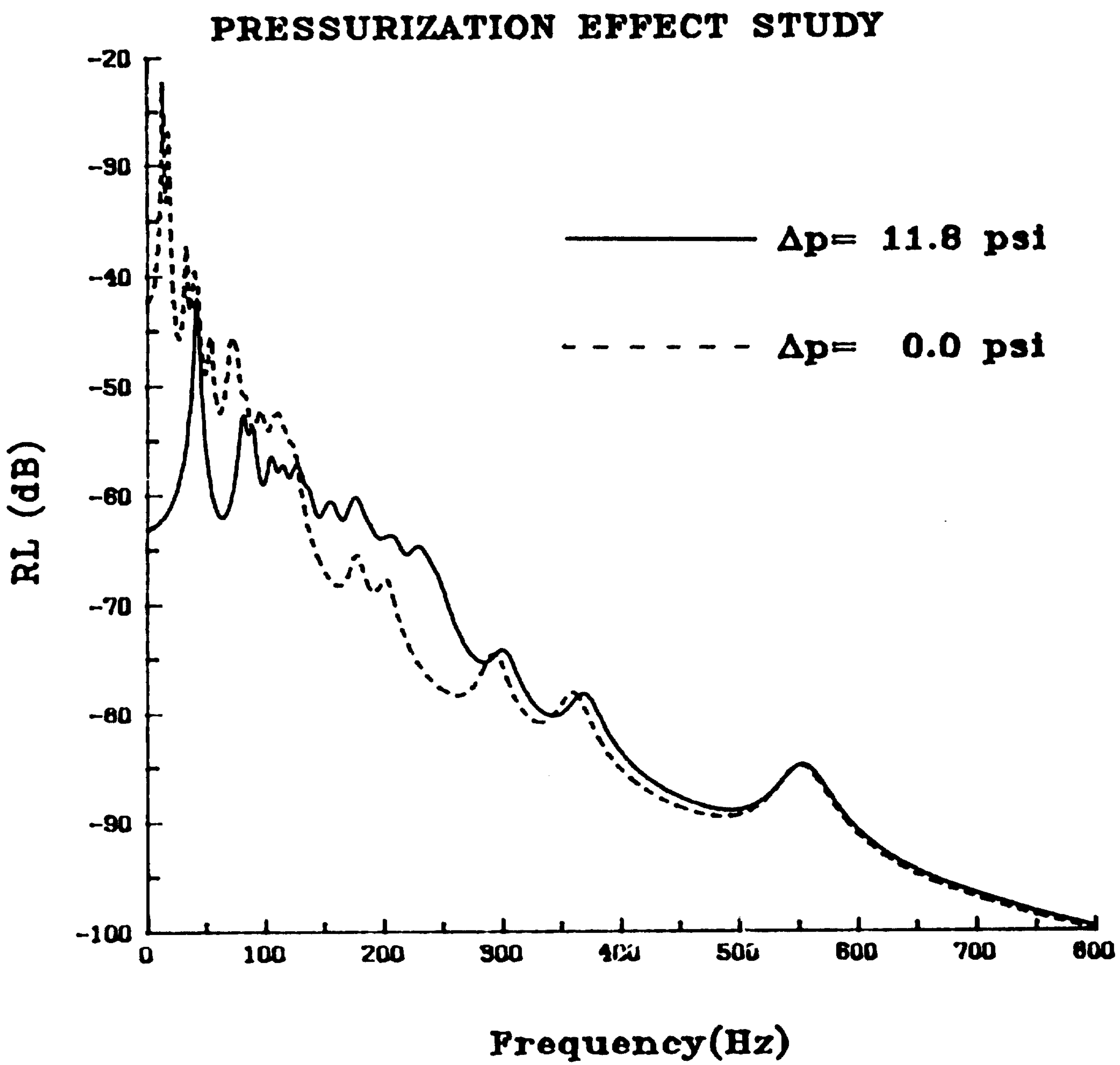

Fig. 4 Deflection Response Levels For Pressurized and Unpressurized Aluminum Shells

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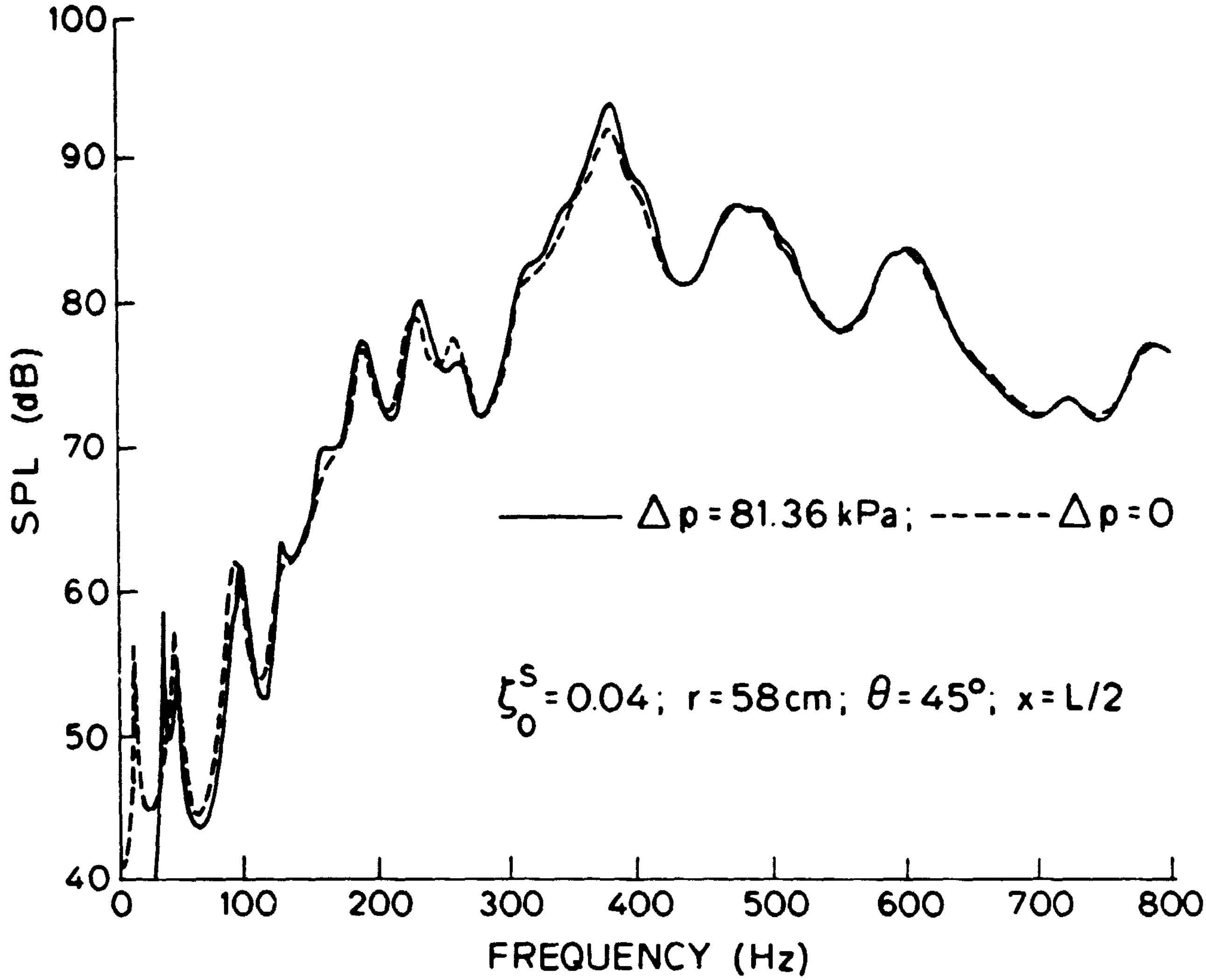

Fig. 5 Sound Transmission Into Pressurized and Unpressurized Shells

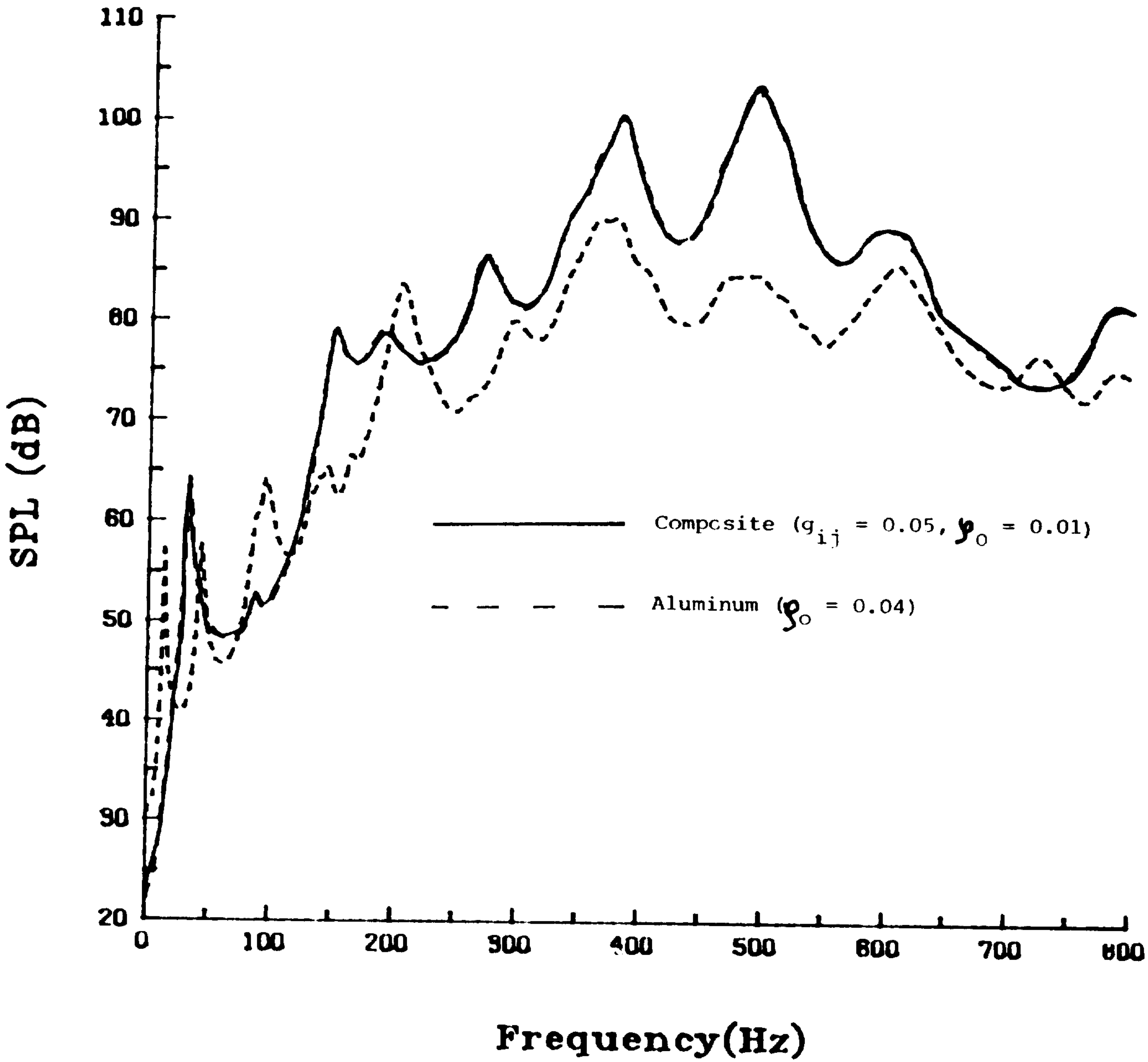

Fig. 6 Sound Pressure Levels for Aluminum and Composite Shells

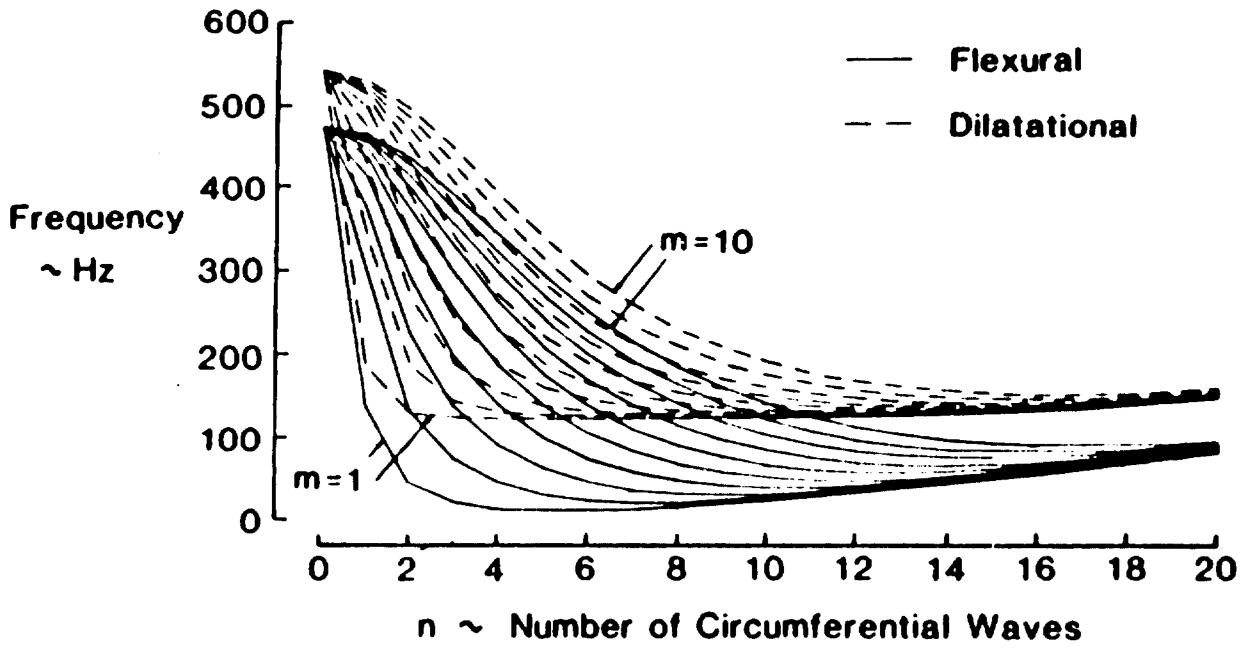

Figure 7. Natural Frequencies for Double Wall Aluminum Shell

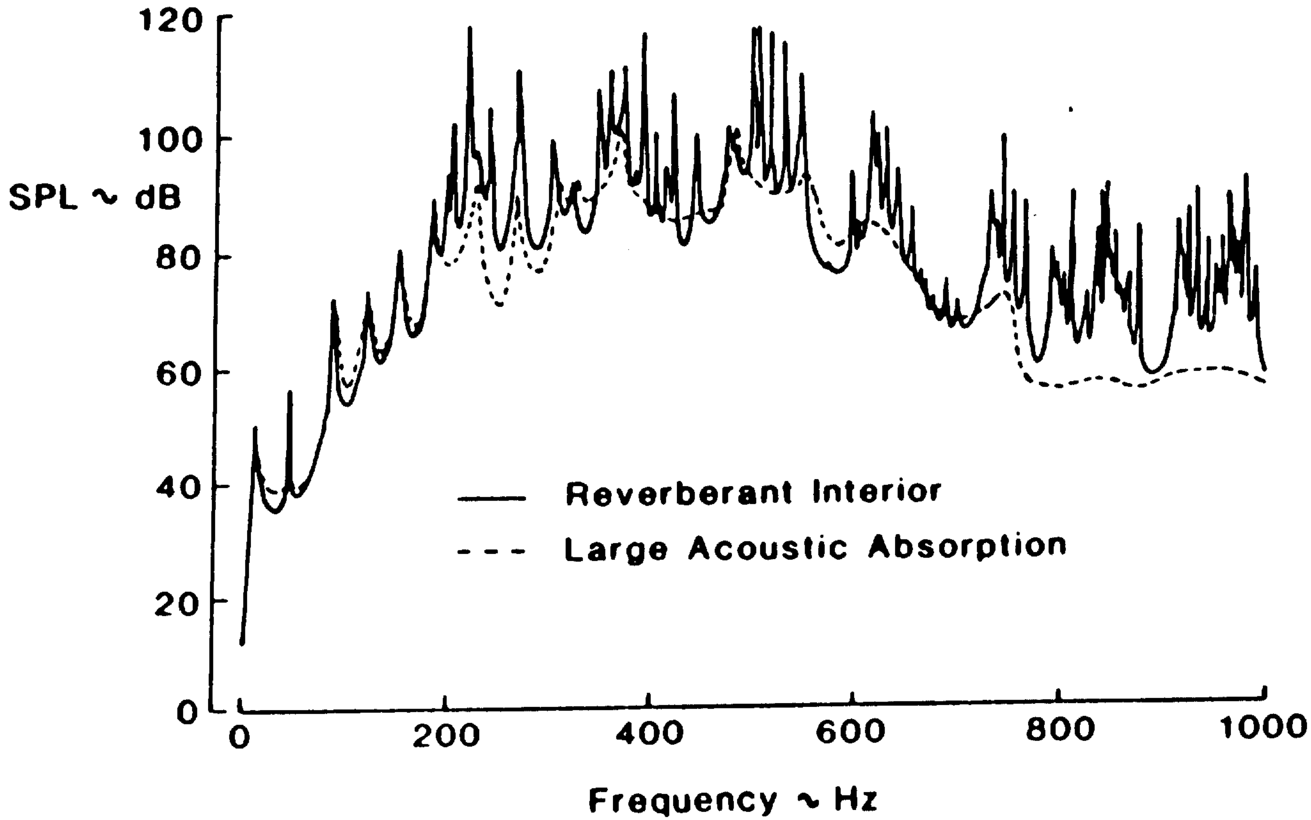

Figure 8. Sound Pressure Levels of a Double Wall Aluminum Shell(Exterior Point Loads)

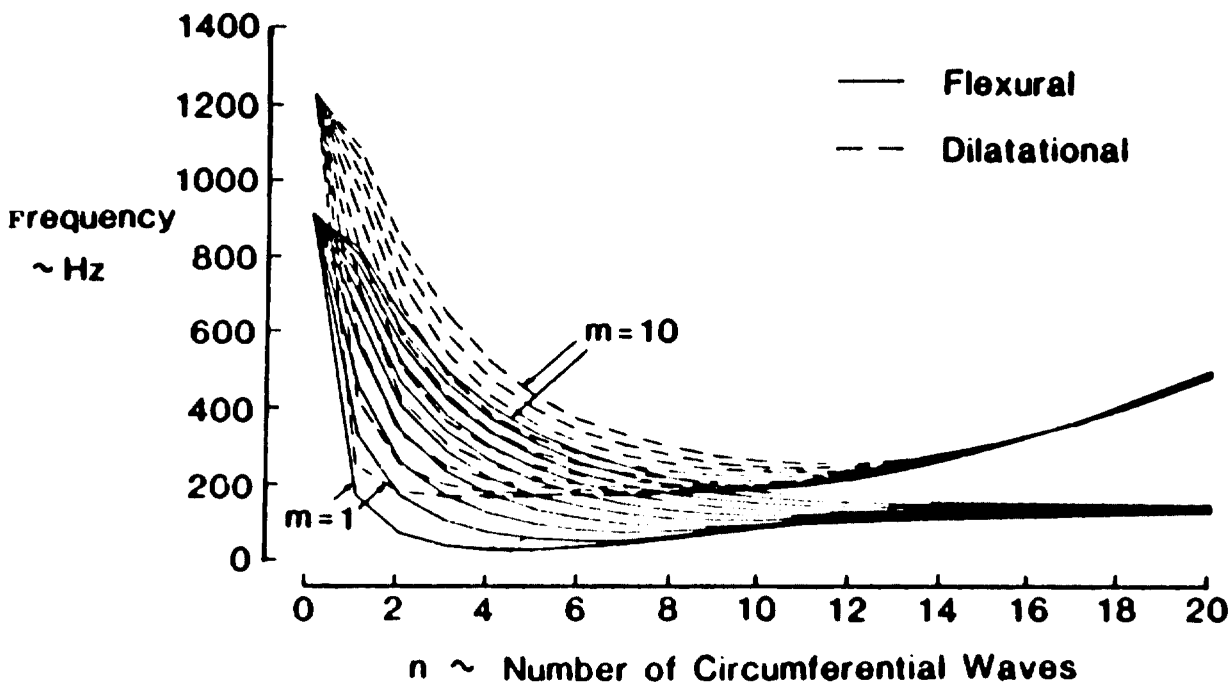

Figure 9. Natural Frequencies for Double Wall Composite Shell

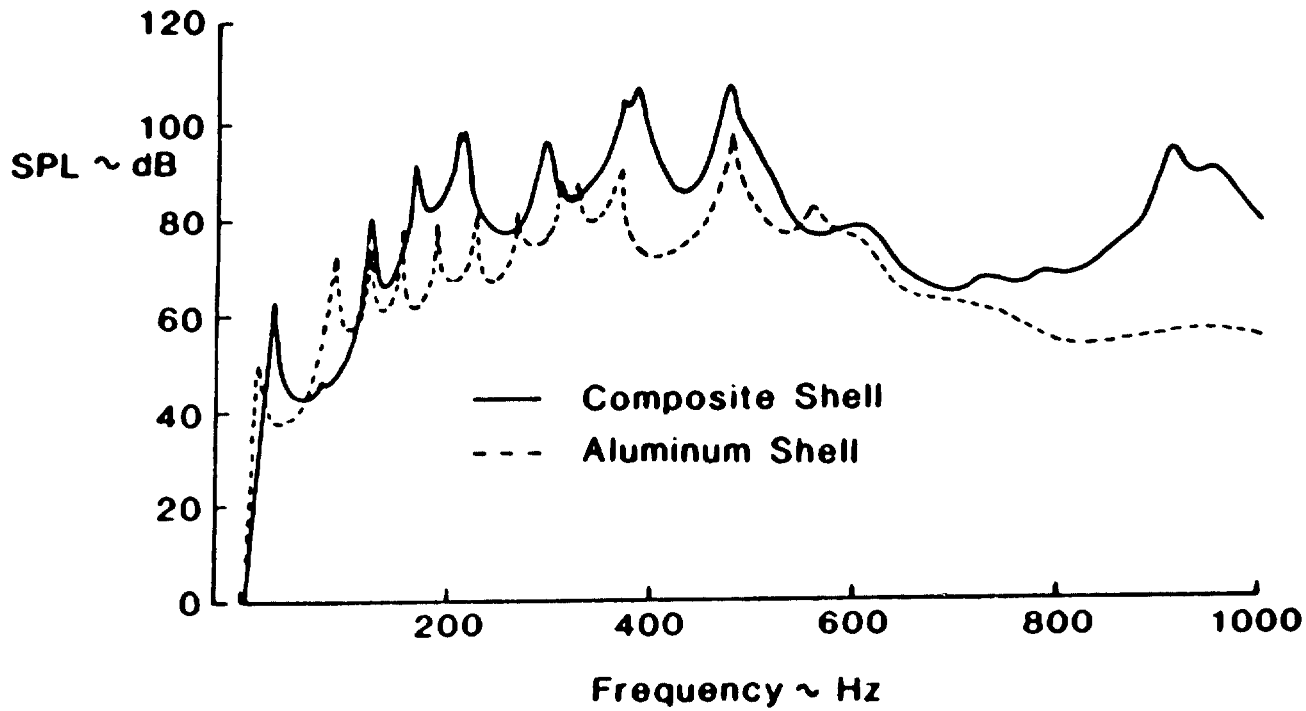

Figure 10. Sound Pressure Levels for Aluminum and Composite Shells(Exterior Point Loads)

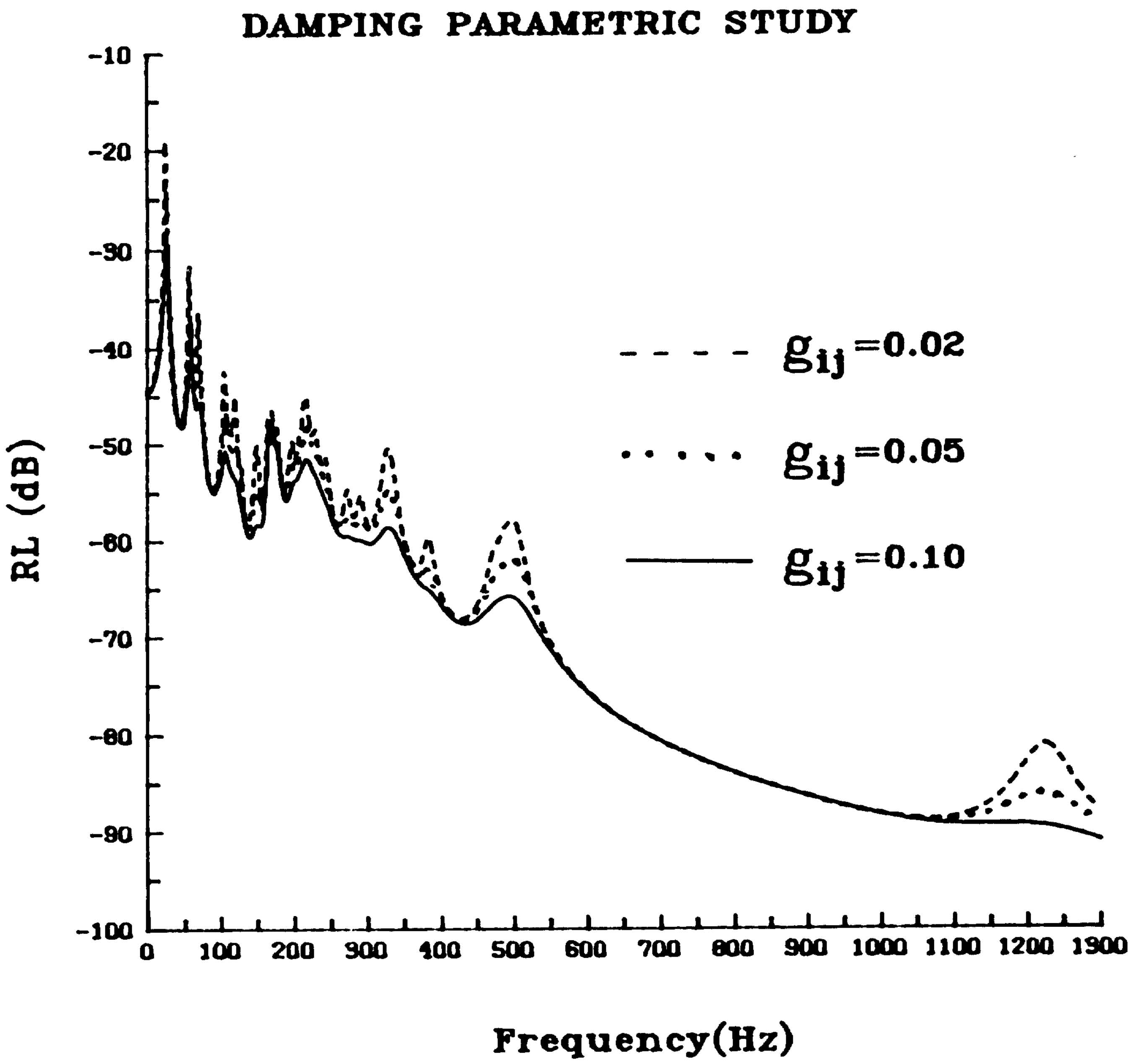

Fig. 11 Deflection response of inner composite shells for different damping loss factor values.

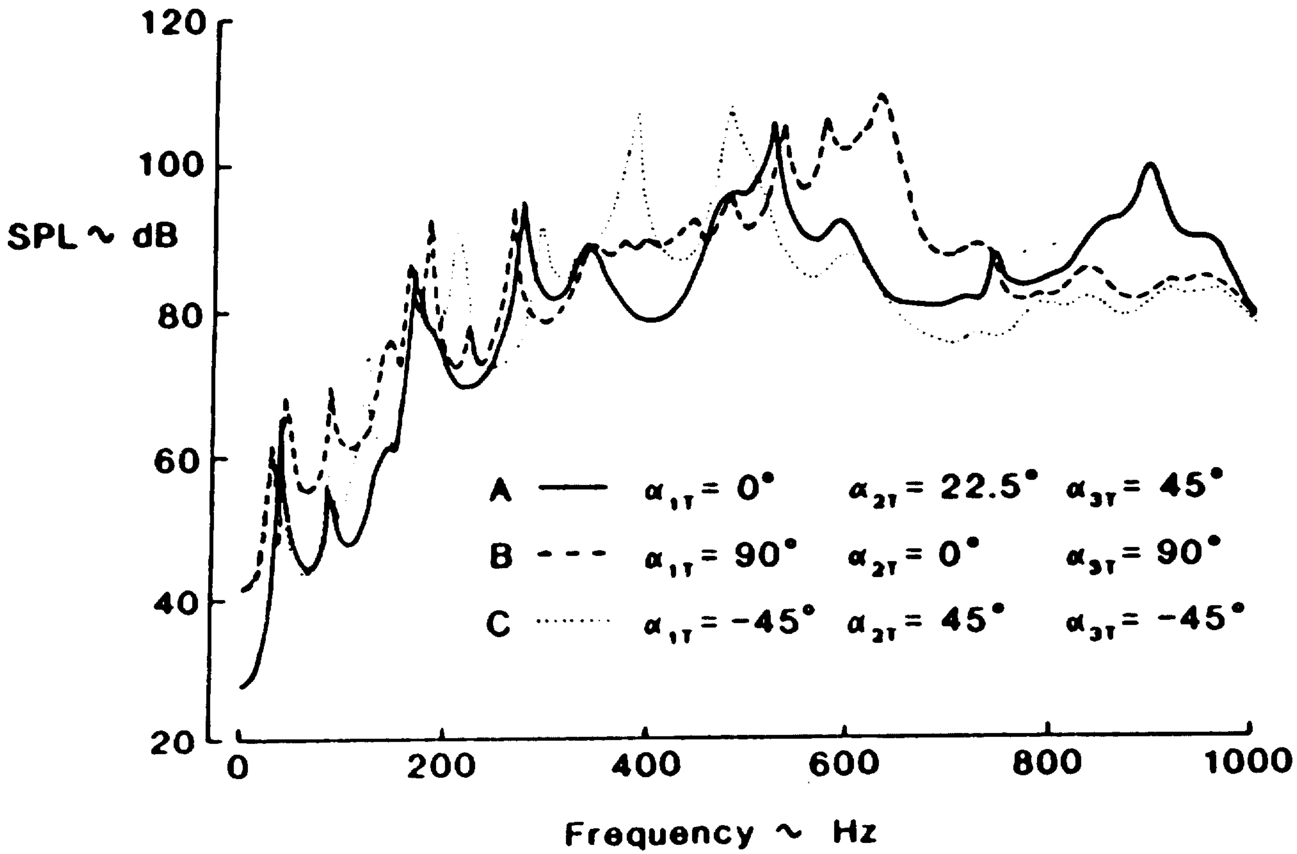

Figure 12. Sound Pressure Levels in a Composite Shell for Different Fiber Orientations(Interior Point Loads)

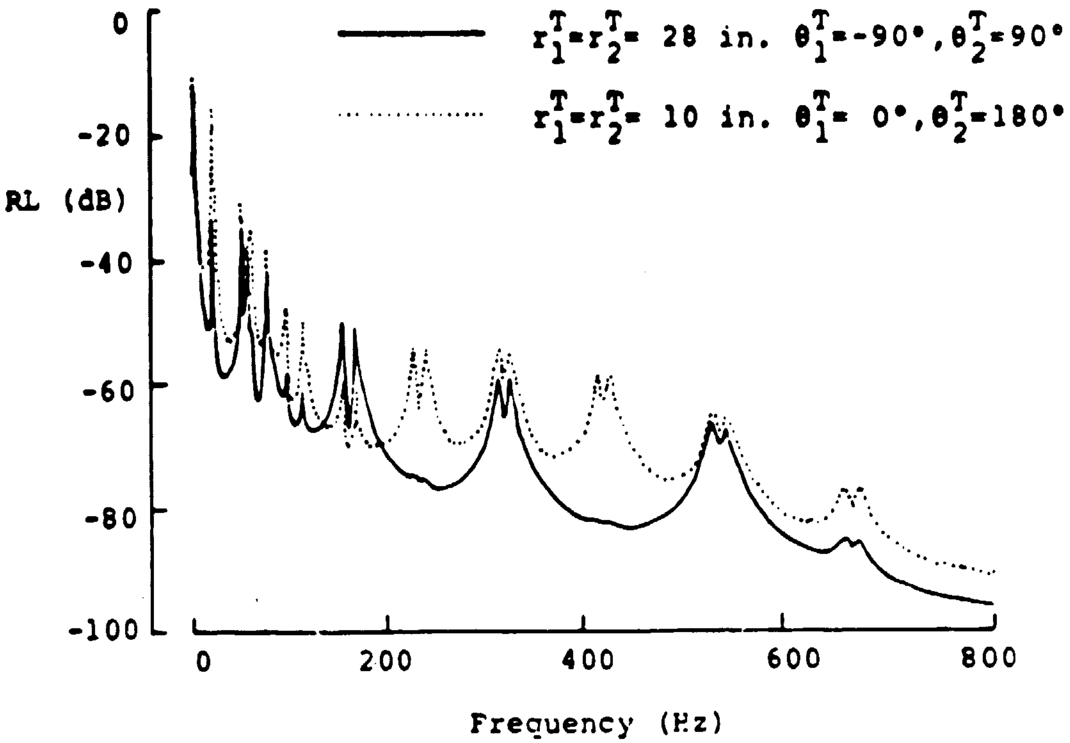

Figure 13. Response Levels of Outer Plate for Different Location of Point Loads(Exterior)

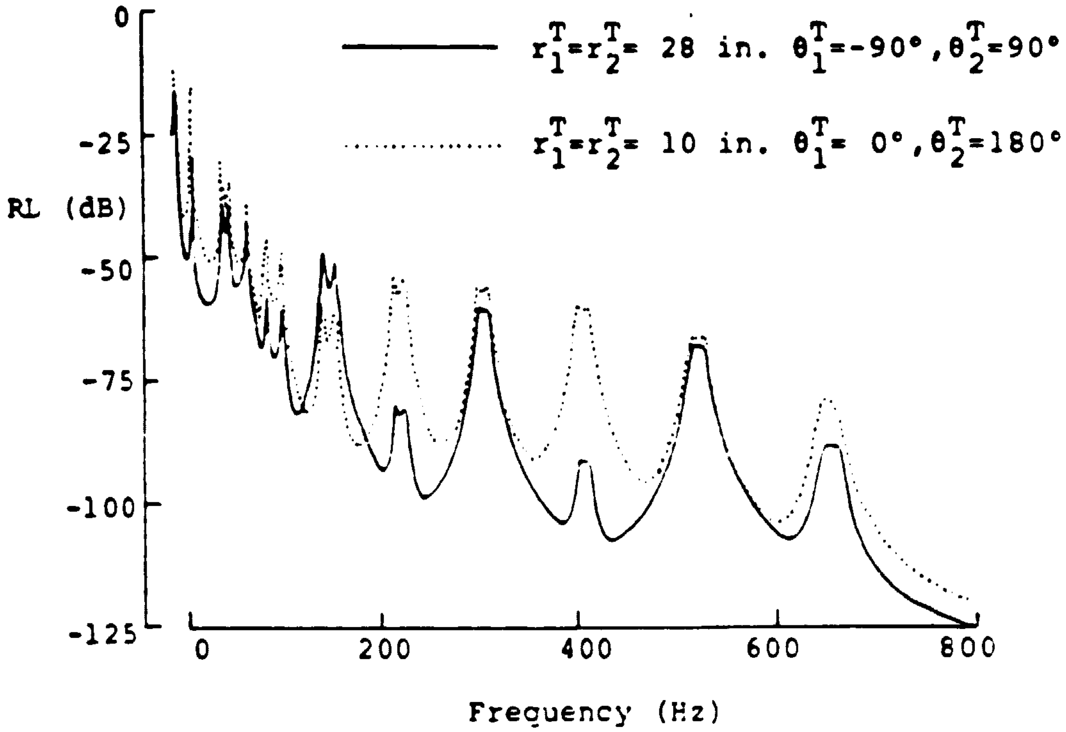

Figure 14. Response Levels of Inner Plate for Different Location of Point Loads(Exterior)

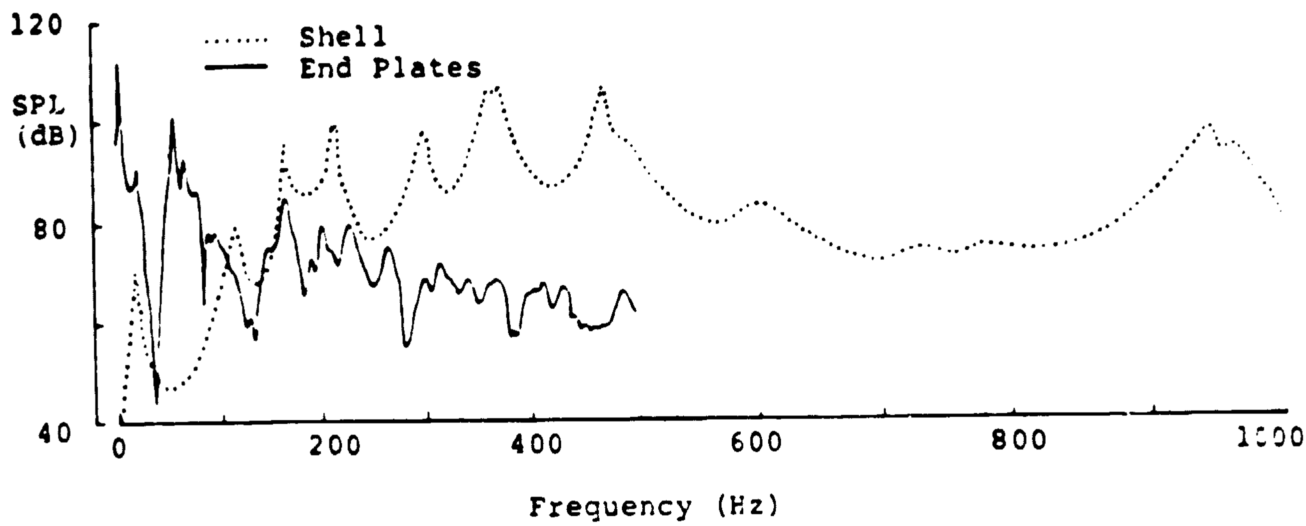

Figure 15. Sound Pressure Levels Due to Individually Vibrating Shell and End Plate Systems (Exterior Point Loads)

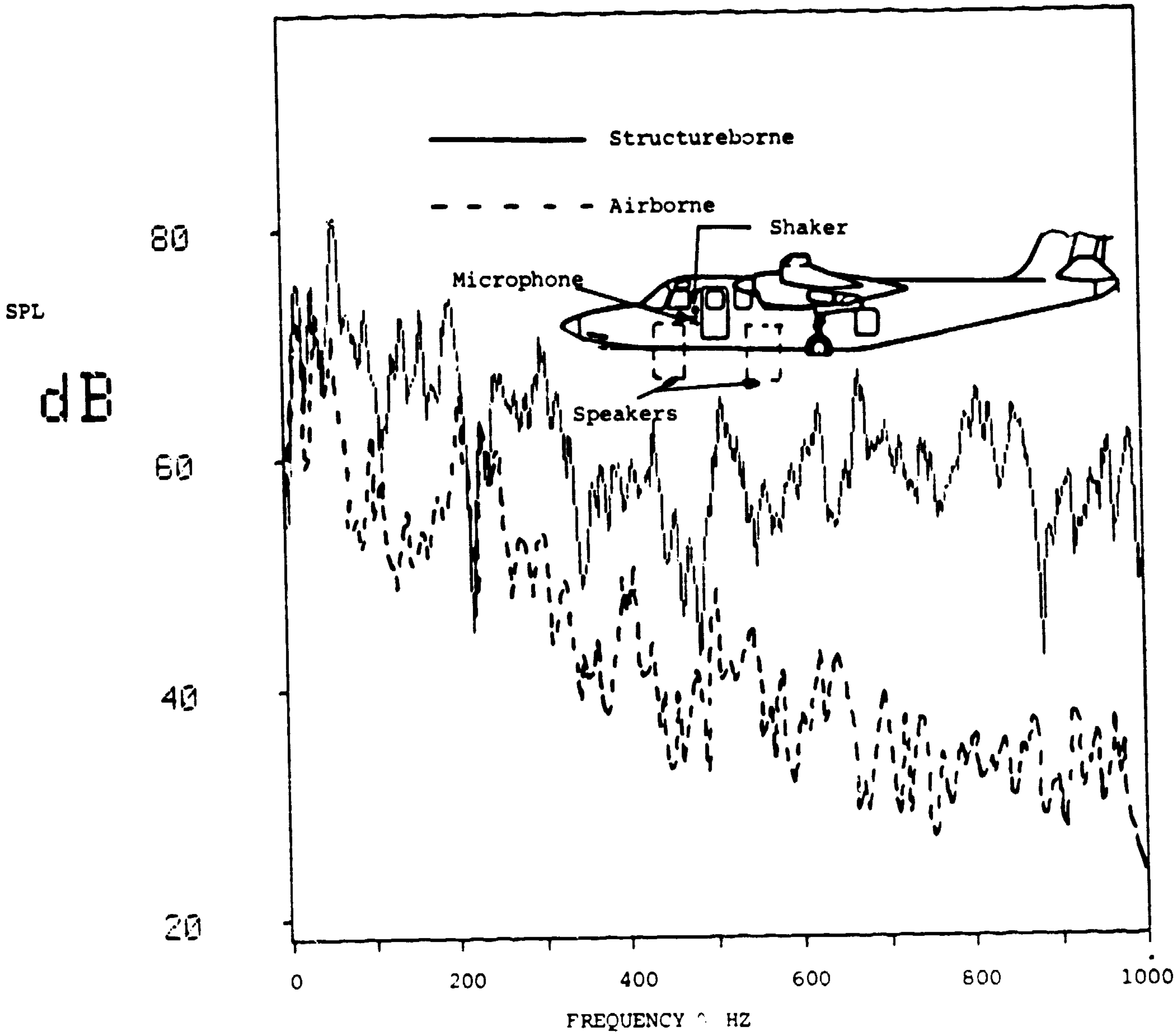

Fig. 16 Airborne and Structureborne Noise Transmission
(Interior Microphone at the source)

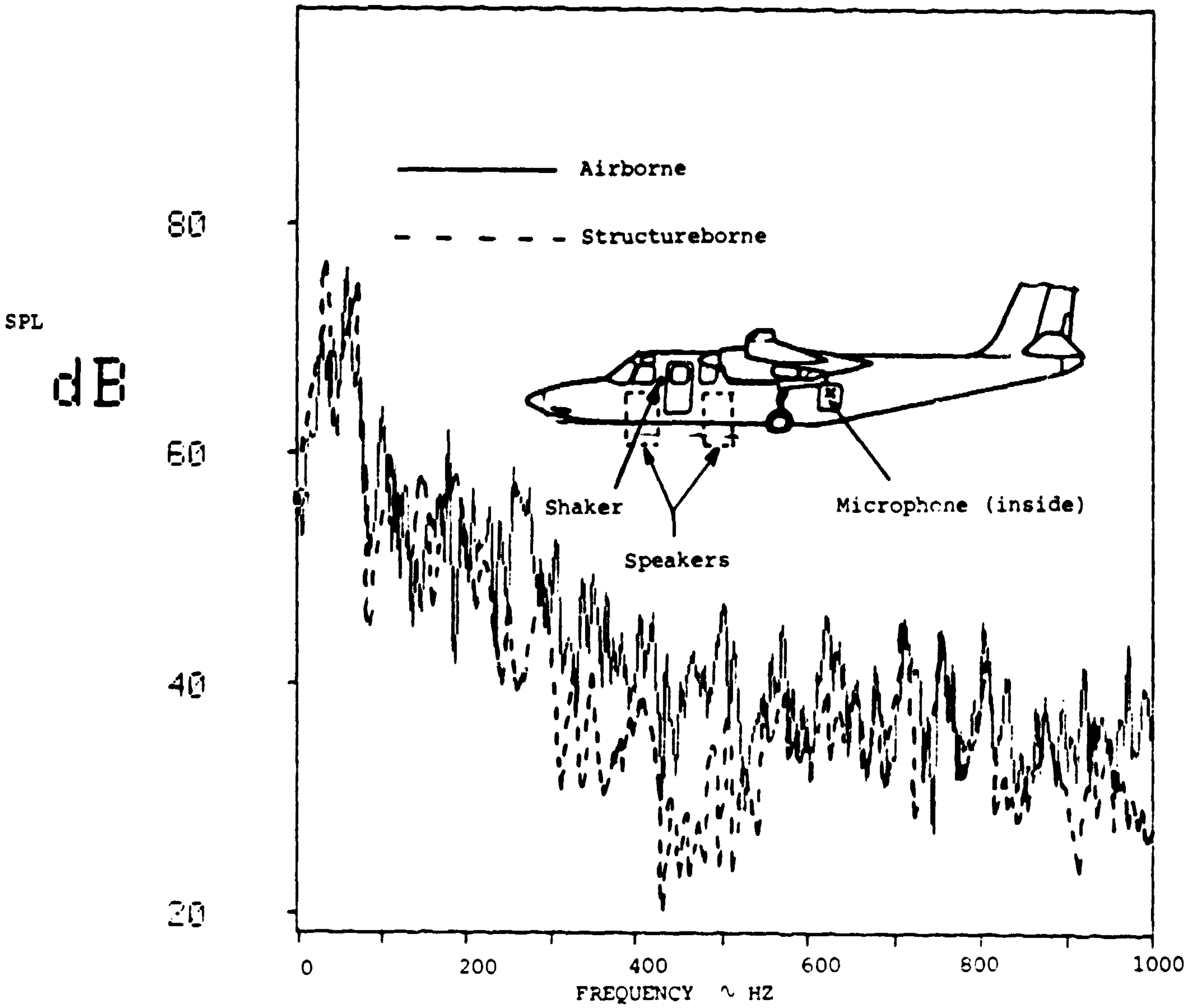

Fig. 17 Airborne and Structureborne Noise Transmission (Interior Microphone About 8Ft. from the source)